

RESEARCH MEMORANDUM

INVESTIGATION OF EFFECTS OF REYNOLDS NUMBER ON LARGE
DOUBLE-ENTRY CENTRIFUGAL COMPRESSOR

By Karl Kovach and Joseph R. Withee, Jr.

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

NATIONAL ADVISORY COMMITTEE
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SUMMARY

An investigation was conducted on a large-capacity double-entry centrifugal compressor to determine the effects of Reynolds number on over-all performance and also to obtain some idea of the origin and nature of the effects. The compressor was operated over a range of inlet conditions equivalent to pressure altitudes of 20,000 to 50,000 feet at equivalent tip speeds of 916, 1309, and 1545 feet per second.

The decreases in the parameters affecting over-all performance at design speed (tip speed of 1545 ft/sec) for a change in pressure altitude from 20,000 to 50,000 feet were: (1) peak total-pressure ratio, 3.4 percent, (2) maximum equivalent weight flow, 3.8 percent, and (3) peak efficiency, 0.02. The origin of at least part of the Reynolds number effects in this compressor is in the inlet and the inducer sections. In the case of a double-entry impeller, the Reynolds number affects not only the matching of the impeller and the diffuser but also the matching of the front and the rear impellers.

INTRODUCTION

The deviations of turbojet-engine performance at altitude from the performance predicted by means of sea-level tests and standard corrections are generally attributed to effects of Reynolds number variation. Several investigations have been made with commercial jet engines submerged in altitude tanks, and the results of these tests have indicated that the performance of the compressor is more affected by Reynolds number than the performance of the other components.

The investigation described in this report was conducted at the NACA Lewis laboratory to obtain quantitative data that show the effects of Reynolds number on the performance of a particular compressor, and also to obtain some insight into the origin and the nature of the effects by observing the behavior of the compressor components as Reynolds number

is varied. This investigation was conducted on a large-capacity double-entry centrifugal compressor. The compressor was instrumented primarily to determine the effects of compressor-component design changes on performance as reported in reference 1. Although the double-entry type compressor does not make an ideal setup for a Reynolds number investigation, it does give sufficient information to show the complex process which results from Reynolds number variation.

The compressor was operated over a range of inlet conditions equivalent to pressure altitudes of 20,000 to 50,000 feet at equivalent tip speeds of 916, 1309, and 1545 feet per second.

APPARATUS

The compressor-flow system consisted of dual inlets containing air-straightening vanes, front and rear inducers, and a two-sided centrifugal impeller that discharged into a vaneless diffuser and then into a vaned diffuser (fig. 1). The impeller-inlet diameter was 18.480 inches and the inlet-to-tip radius ratio was 0.649. The impeller had 23 straight radial blades and the inducer had 23 parabolic blades. The vaned diffuser had 14 passages; the area at the throat of each passage was 6.720 square inches.

A steel tank 6 feet in diameter and $13\frac{1}{2}$ feet long housed the compressor assembly. This tank was fitted with three sets of screens to insure uniform flow into the compressor. Air was discharged through 14 outlet ducts into the laboratory exhaust system.

The compressor was driven by a 9000-horsepower variable-frequency electric motor through a geared speed increaser.

INSTRUMENTATION

Weight-flow measurements were made with a submerged adjustable orifice or a submerged flat-plate orifice depending on the magnitude of the flow.

Instrumentation was installed in the depression tank of the inlet according to the standards set up in reference 2. The state of the outlet air was determined from measurements taken in each of the 14 outlet ducts. The instrumentation in each duct consisted of one thermocouple, two total-pressure probes, and two wall static-pressure taps.

There were 100 wall static-pressure taps located on the front and rear impeller casings and on all four walls of one of the vaned diffuser passages to give complete instrumentation of an air-flow path through the compressor.

A remotely controlled cylindrical yaw tube was installed at the inlet to one of the vaned diffuser passages to measure the total-pressure gradient across the passage.

Pressures were measured by use of water and mercury manometers. All temperatures were measured with a calibrated potentiometer and a spot-light galvanometer. The compressor speed was measured by an electric tachometer.

The measurements are estimated to be accurate within the following limits:

Temperature, °F	±0.5
Pressure, in. Hg abs	±0.04
Air weight flow, percent	±1.0
Impeller speed, percent	±0.5

On the basis of the accuracy of these measurements, the accuracy of the data at design speed is within the following limits:

Compressor total-pressure ratio, percent	±0.3
Compressor adiabatic temperature-rise efficiency	±0.006

PROCEDURE

All runs were made at ambient inlet temperatures which varied from 60° to 85° F. The compressor was operated at the following conditions:

Equivalent impeller speed $N/\sqrt{\theta}$ (rpm)	Equivalent tip speed $U/\sqrt{\theta}$ (ft/sec)	Inlet total pressure P_1 (in. Hg abs)
7,000	916	6, 10, 14, 25
10,000	1309	5, 10, 14, 20
11,800 (design)	1545	5, 10, 14, 18

Total-pressure surveys at the vaned diffuser inlet were made at an equivalent impeller tip speed of 1309 feet per second and inlet pressures of 5 and 14 inches of mercury absolute.

The range of weight flows investigated in all cases was from maximum to the point of incipient surge.

COMPUTATIONS

Computations of adiabatic temperature-rise efficiency η_{ad} for the compressor were made in accordance with reference 3. The equivalent weight-flow parameter $W\sqrt{\theta}/\delta$ and the equivalent speed parameter $N/\sqrt{\theta}$ were computed according to the method of reference 2. All symbols are defined in appendix A, and the method of computing effective area variation at the choking point in the vaned diffuser is derived in appendix B. The Reynolds number index $P_1/\mu_1\sqrt{T_1}$ is based on the assumption that the axial Mach number is constant at the impeller inlet for constant equivalent impeller speed and equivalent weight flow. The velocity of the air at the inlet is therefore proportional to the square root of the temperature. For comparative purposes the values of the gas constant and the length parameter may be omitted so the Reynolds number is directly proportional to the pressure and inversely proportional to the viscosity and the square root of the temperature. The relation between equivalent altitude and Reynolds number index was found by computing the Reynolds number index from the NACA standard temperature and pressure for each altitude.

RESULTS AND DISCUSSION

Over-All Performance

Inlet pressures of 14 inches of mercury absolute and lower. - The variation in peak compressor performance with Reynolds number index is shown in figure 2 for four similar double-entry compressors, each at design speed. Although the magnitudes of the performance changes are not the same for all four compressors the trend is definitely the same, that is, increasing performance with increasing Reynolds number (decreasing altitude). Figure 3 shows the variations of compressor total-pressure ratio and adiabatic temperature-rise efficiency with equivalent weight flow for several inlet pressures and equivalent tip speeds of 916, 1309, and 1545 feet per second for the 23-blade compressor which is discussed in this report. The variation of the peak performance of this compressor with Reynolds number index is summarized for the three equivalent tip speeds in figure 4. For a change in Reynolds number index from about 5.0×10^4 to 1.75×10^4 (corresponding to a change in pressure altitude from about 20,000 to 50,000 ft) at the design equivalent tip speed of 1545 feet per second, the compressor performance decreased as follows: (1) peak total-pressure ratio, 3.4 percent, (2) maximum equivalent weight flow, 3.8 percent, and (3) peak efficiency, 0.02. The rate of decrease in performance became greater as the inlet pressure (and therefore Reynolds number index) was decreased.

Inlet pressures of 14 inches of mercury absolute and higher. - Compressor performance for operation at inlet pressures of 14 inches of mercury absolute and higher are presented in figure 5 for equivalent tip speeds of 916, 1309, and 1545 feet per second. No appreciable increase in performance occurred at any speed for the higher inlet pressures; in fact, for speeds of 1309 and 1545 feet per second, there appears to be a slight decrease in total-pressure ratio and adiabatic efficiency (figs. 5(c) to 5(f)). Previous investigations of Reynolds number effects on engine performance have also shown that, below equivalent pressure altitudes of approximately 25,000 feet (inlet pressure of 14 in. Hg abs), there is no measurable change in performance with change in Reynolds number.

Static-Pressure Distribution Along Inlet Wall

Since the Reynolds number within a centrifugal impeller increases rapidly from inlet to outlet and since a small pressure loss at the inlet results in a large change in over-all performance, the most logical place to look for the origin of Reynolds number effects appears to be the compressor inlet and the inducer sections. A sketch of the front and rear inlets is presented in figure 1; the shape of the air-flow path, the locations of the static-pressure taps, and the hot outlet ducts, which the air entering the rear inlet has to pass over, are shown in this figure. Data are not presented for static taps 1, 2, 5, and 6 because the velocities at these stations were so low that trends could not be observed. The static pressures measured at static taps 3 and 4 (see fig. 1) on the front inlet shroud at design equivalent tip speed, 1545 feet per second, and two inlet pressures, 5 and 14 inches of mercury absolute, are presented in figure 6(a) as a function of equivalent weight flow. For equal equivalent flow rates over the entire flow range, the static-pressure ratios for the runs at an inlet pressure of 5 inches of mercury are lower than the static pressure ratios for the runs at an inlet pressure of 14 inches of mercury; this indicates that the velocities are higher for the lower inlet pressure. The effective local flow areas must therefore have been decreased by a change in the boundary-layer formation. Since the only change in flow conditions upstream of static taps 3 and 4 was the inlet pressure, this change in boundary layer must result from the lower value of Reynolds number. The origin of at least some of the Reynolds number effects on over-all performance must therefore be in the compressor-inlet and inducer sections. Data for the same operating conditions are shown in figure 6(b) for the rear static taps 7 and 8. The same trends occurred at the rear inlet as at the front inlet. It can also be noted in figure 6 that the viscous effects are greater at the static taps closest to the impeller; this is shown by the greater spread in static-pressure ratio between the 5-inch and the 14-inch inlet-pressure runs at static taps 4 and 8 than at taps 3 and 7.

The variation of static-pressure ratio with equivalent weight flow for inlet pressures of 5 and 14 inches of mercury is also presented for an equivalent tip speed of 1309 feet per second. Figure 7(a) is for the front inlet and figure 7(b) is for the rear inlet. At a tip speed of 1309 feet per second, the static-pressure ratios for the 5-inch inlet-pressure runs became increasingly lower than the static-pressure ratios for the 14-inch runs as the air progressed toward the impeller. This trend is apparent only at the high weight flows because of discontinuities in the curves that occur at a weight flow of about 81.5 pounds per second for the 14-inch inlet-pressure runs and at about 85 pounds per second for the 5-inch inlet-pressure runs.

These discontinuities are caused by an abrupt redistribution of the flow between the front and the rear impellers. The conditions at the inlet to the front and the rear impellers are not identical because the air entering the rear inlet becomes heated in passing over the hot outlet ducts (fig. 1). The result is that the two halves of the system are actually operating at different equivalent speeds and different equivalent weight flows. Both sides of the impeller do, however, discharge into a common diffuser. The impeller operation therefore shifts back and forth on the individual operating curves until a balance in static pressure is attained at the outlet.

Examination of the data indicated that a flow shift occurred at speeds lower than design, but at design speed the compressor surged violently before the redistribution of air between the front and rear impellers was completed. Figure 7 also indicates that the Reynolds number affects the severity of the weight flow shift; that is, the lower the Reynolds number the less severe the flow shift.

Total-Pressure Surveys at Vaned-Diffuser Inlet

The results of total-pressure surveys made at the vaned-diffuser inlet at an equivalent tip speed of 1309 feet per second for inlet pressures of 5 and 14 inches of mercury absolute are presented in figure 8. Curves for two flow conditions are presented: (1) peak compressor pressure ratio at each inlet condition, figure 8(a), and (2) constant equivalent weight flow corresponding to the maximum at an inlet pressure of 5 inches of mercury absolute, figure 8(b). For both flow conditions, the total-pressure ratio is higher at the higher Reynolds number; this indicates that the primary effect of Reynolds number on compressor total-pressure ratio occurs upstream of the vaned diffuser. Actually, appreciable effects resulting from changes in Reynolds number alone were not expected in the diffuser because of the high values of Reynolds number in this region. Since these surveys were made downstream of the vaneless diffuser where the mixing losses occur, there is no simple relation between the shape of these curves and the distribution of flow between the front and the rear impellers.

Effective Flow Area in Vaned Diffuser

The variation of effective flow area at the choking point in the vaned diffuser with Reynolds number index is presented in figure 9 for equivalent tip speeds of 1309 and 1545 feet per second. There is a definite increase in effective area with increasing values of Reynolds number index. This area increase is probably the result of flow directions at the diffuser inlet, which are closer to the design conditions, rather than differences in boundary layer in the diffuser. The increase in the effective flow area together with the increased pressure ratio delivered by the impeller accounts for the increase in compressor air-flow rate with Reynolds number index noted previously.

SUMMARY OF RESULTS

An investigation of the effects of Reynolds number on a large-capacity, double-entry centrifugal compressor produced the following results:

1. The decreases in over-all performance at design speed for a change in pressure altitude from 20,000 to 50,000 feet were: (1) peak total-pressure ratio, 3.4 percent; (2) maximum equivalent weight flow, 3.8 percent; and (3) peak efficiency, 0.02.

2. The origin of at least part of the Reynolds number effects in this compressor are in the inlet and the inducer sections.

3. The Reynolds number affects the severity of the redistribution of weight flow between the front and the rear impellers at speeds lower than design.

4. The primary effect of Reynolds number on total-pressure ratio occurs upstream of the vaned diffuser.

5. The effective area at the choking point in the vaned diffuser increased with increasing Reynolds number.

6. There was no appreciable change in compressor performance at Reynolds numbers corresponding to inlet pressures higher than 14 inches of mercury absolute (equivalent pressure altitude, 25,000 ft).

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio

APPENDIX A

SYMBOLS

The following symbols are used in this report:

A	effective critical diffuser throat area, sq ft
g	acceleration due to gravity, 32.2 ft/sec
M	Mach number
N	impeller speed, rpm
P	total pressure, in. Hg abs
p	static pressure, in. Hg abs
R	universal gas constant, 53.4 ft-lb/(lb)(°R)
T	total temperature, °R
t	static temperature, °R
U	impeller tip speed, ft/sec
V	air velocity, ft/sec
W	compressor weight flow, lb/sec
γ	ratio of specific heats
δ	ratio of inlet total pressure to NACA standard sea-level pressure, P/29.92
η_{ad}	adiabatic temperature-rise efficiency
θ	ratio of inlet total temperature to NACA standard sea-level tem- perature, T/519
μ	absolute air viscosity, lb/ft-sec
ρ	air density, lb/cu ft

Subscripts:

- 1 compressor inlet
- 2 vaned diffuser inlet
- 3 compressor outlet

Superscripts:

- ' conditions at specified inlet pressure
- " conditions at inlet pressure of 14 in. Hg abs

APPENDIX B

METHOD OF COMPUTING EFFECTIVE AREA RATIO AT CHOKING POINT IN

VANED DIFFUSER

The continuity equation for compressible flow is

$$\rho VA = W$$

If the continuity equation is rearranged, if the density is expressed in terms of temperature and pressure, and if the velocity is given in terms of Mach number, then

$$A = \frac{W}{\rho V} = \frac{WRt}{pM \sqrt{\gamma g R t}}$$

When appropriate subscripts and superscripts are used, the following ratio can be written:

$$\frac{A_2'}{A_2''} = \frac{\frac{W_2' R_2' t_2'}{p_2' M_2' \sqrt{\gamma g R_2' t_2'}}}{\frac{W_2'' R_2'' t_2''}{p_2'' M_2'' \sqrt{\gamma g R_2'' t_2''}}} = \frac{W_2'}{W_2''} \sqrt{\frac{t_2'}{t_2''}} \frac{p_2''}{p_2'} \frac{M_2''}{M_2'}$$

Since at the choking point in the diffuser $M_2' = M_2'' = 1$,

$$\frac{t_2'}{t_2''} = \frac{T_2'}{T_2''}$$

therefore

$$\frac{A_2'}{A_2''} = \frac{W_2'}{W_2''} \frac{p_2''}{p_2'} \sqrt{\frac{T_2'}{T_2''}}$$

The quantities W_2' and W_2'' are measured and p_2' and p_2'' are obtained from wall static taps at the minimum-area point in the vaned diffuser. The total temperature is assumed to remain constant through the diffuser so that T_2' and T_2'' are obtained from measurements at the compressor-outlet measuring station.

REFERENCES

1. Withee, Joseph R., Jr., Kovach, Karl, and Ginsburg, Ambrose: Experimental Investigation of Effects of Design Changes on Performance of Large-Capacity Centrifugal Compressors. NACA RM E50K10, 1951.
2. NACA Subcommittee on Compressors: Standard Procedures for Rating and Testing Multistage Axial-Flow Compressors. NACA TN 1138, 1946.
3. Ellerbrock, Herman H., Jr., and Goldstein, Arthur W.: Principles and Methods of Rating and Testing Centrifugal Superchargers. NACA ARR, Feb. 1942.

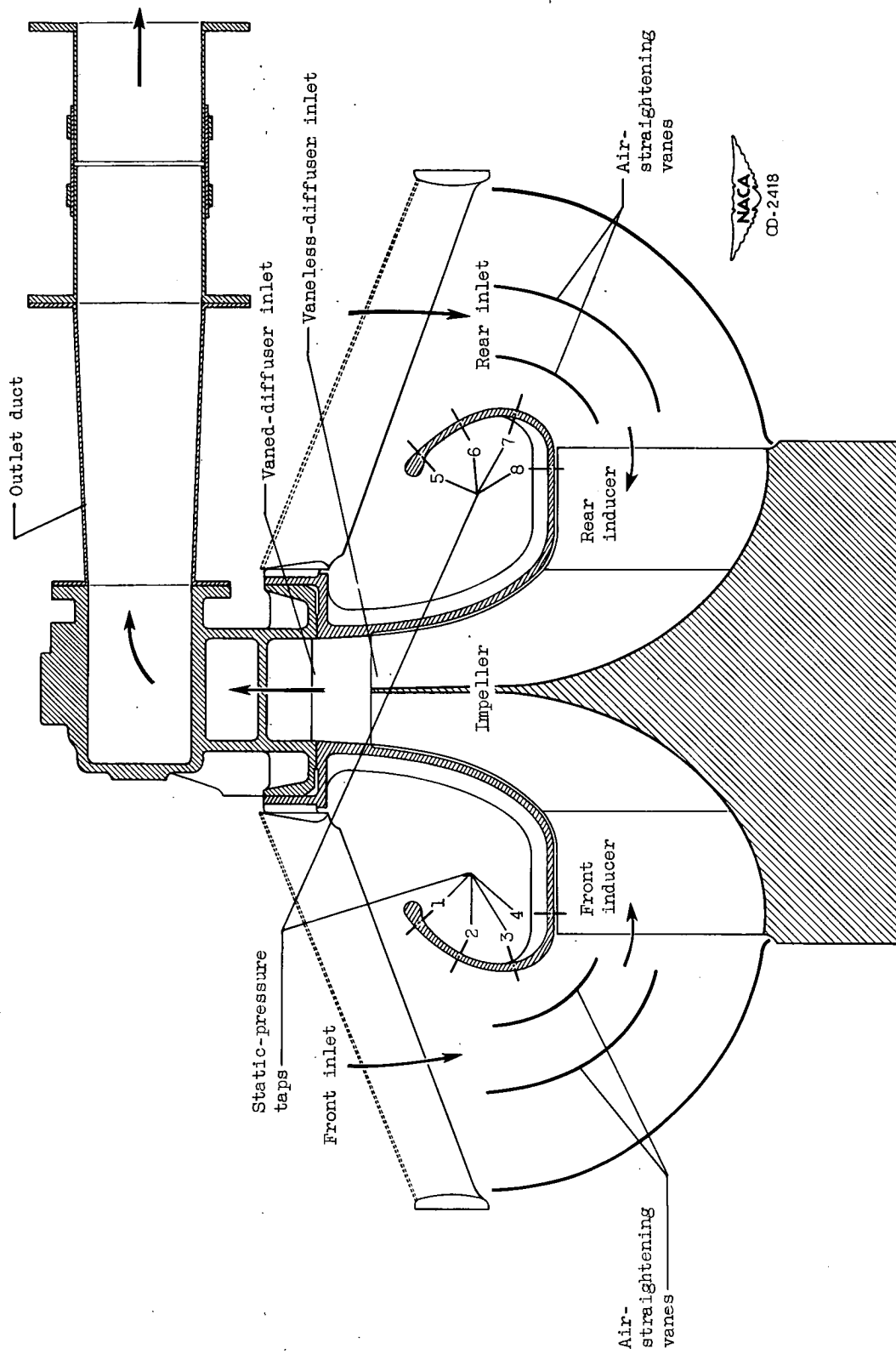


Figure 1. - Air-flow path through compressor.

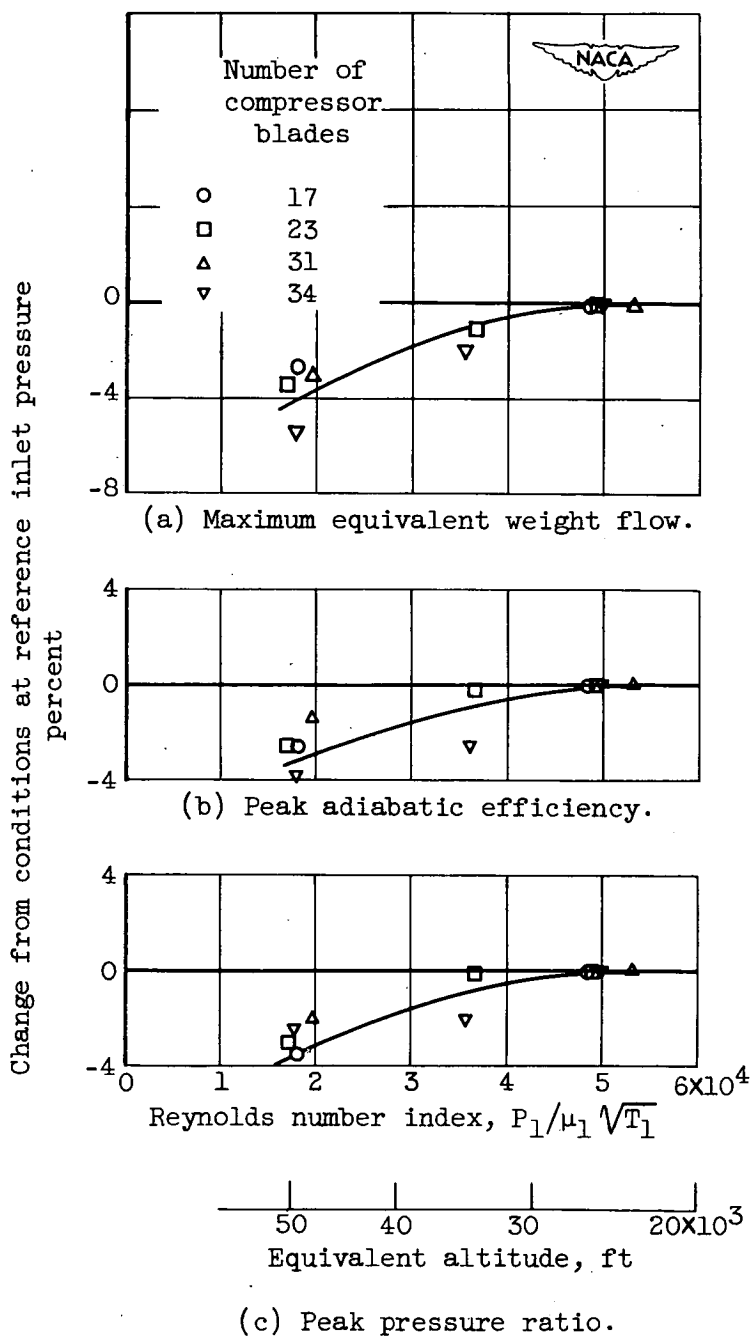


Figure 2. - Variation of peak compressor performance with Reynolds number index for four similar double-entry centrifugal compressors, each at design speed. Reference inlet pressure, 14 inches of mercury absolute.

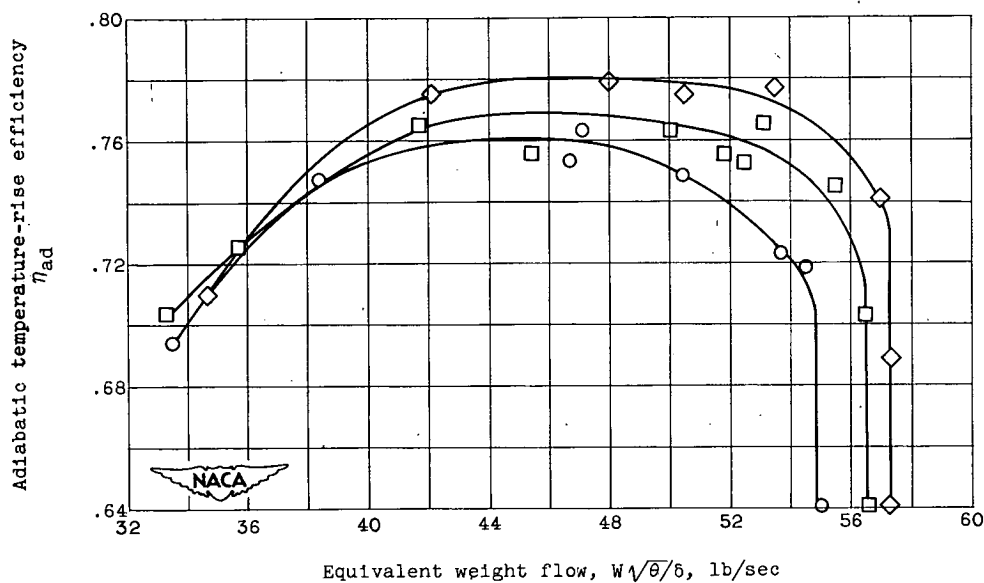
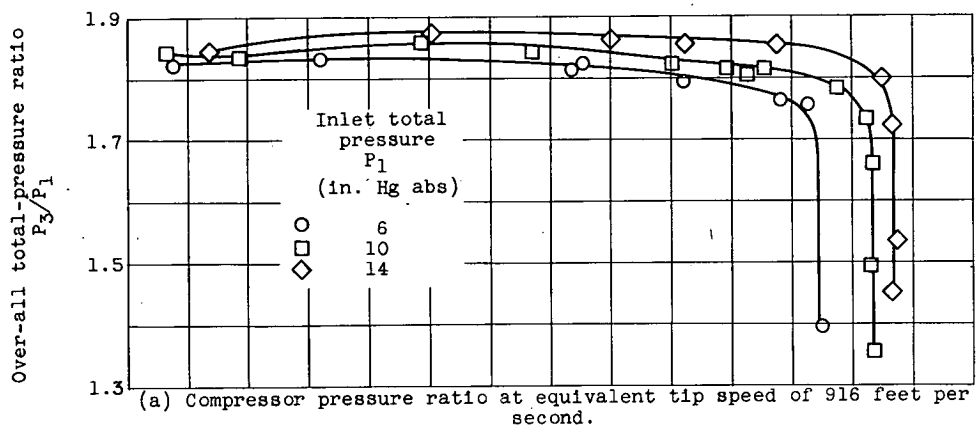


Figure 3. - Over-all performance of 23-blade compressor at four inlet pressures.

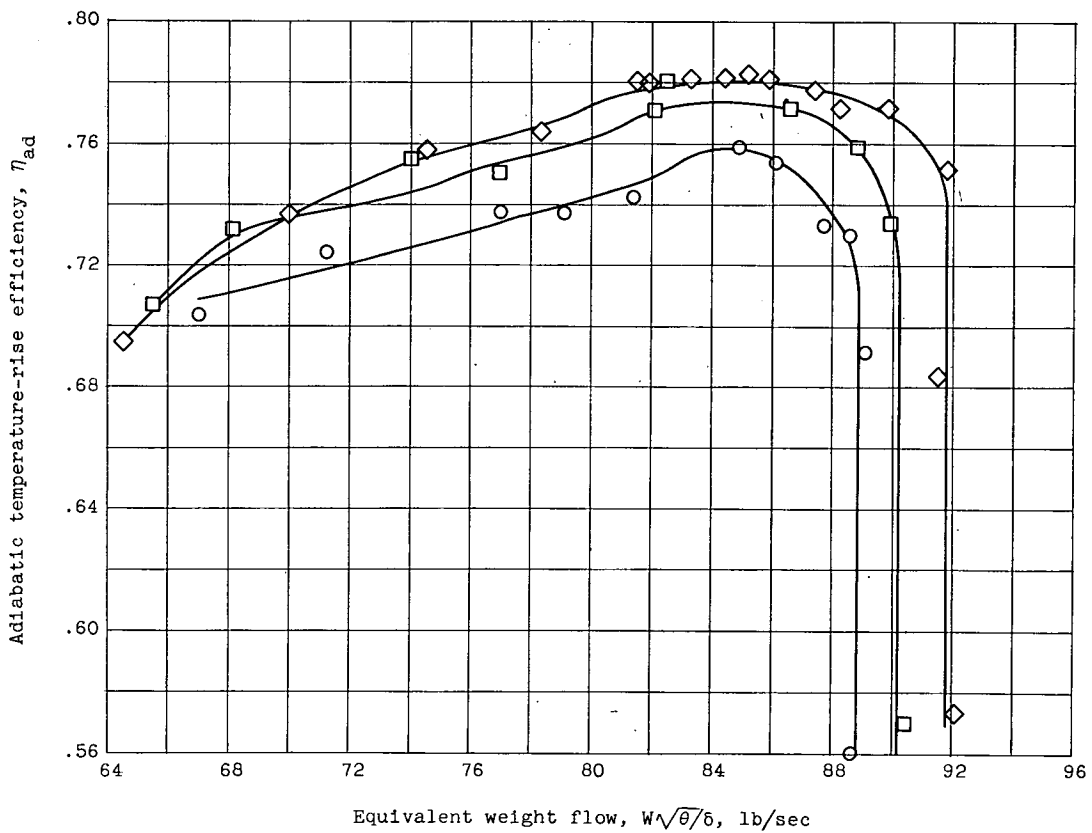
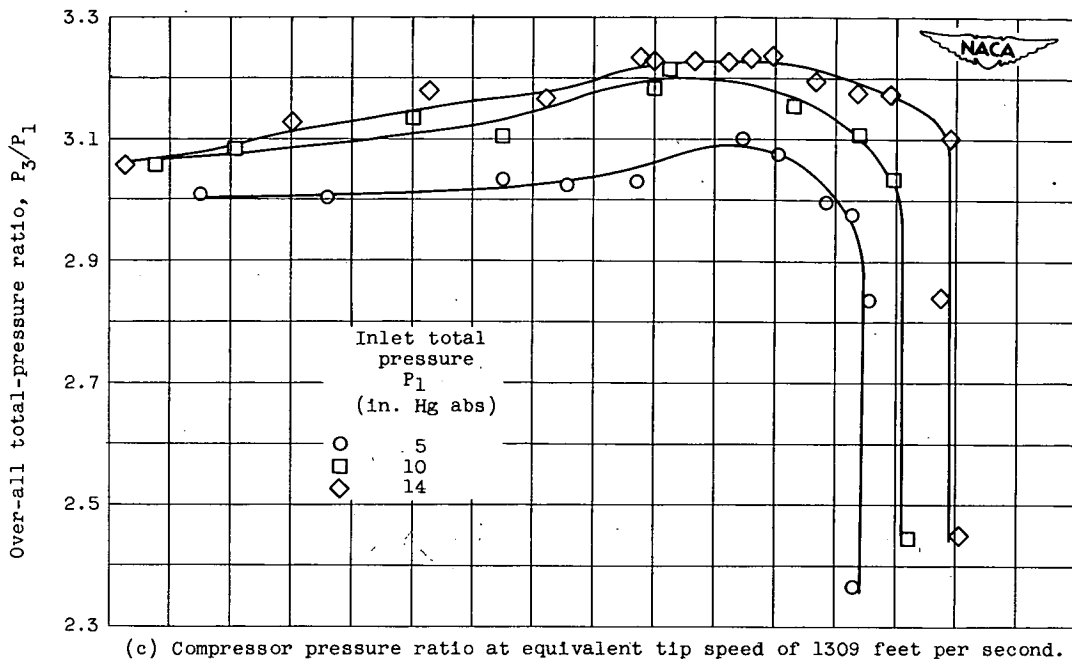
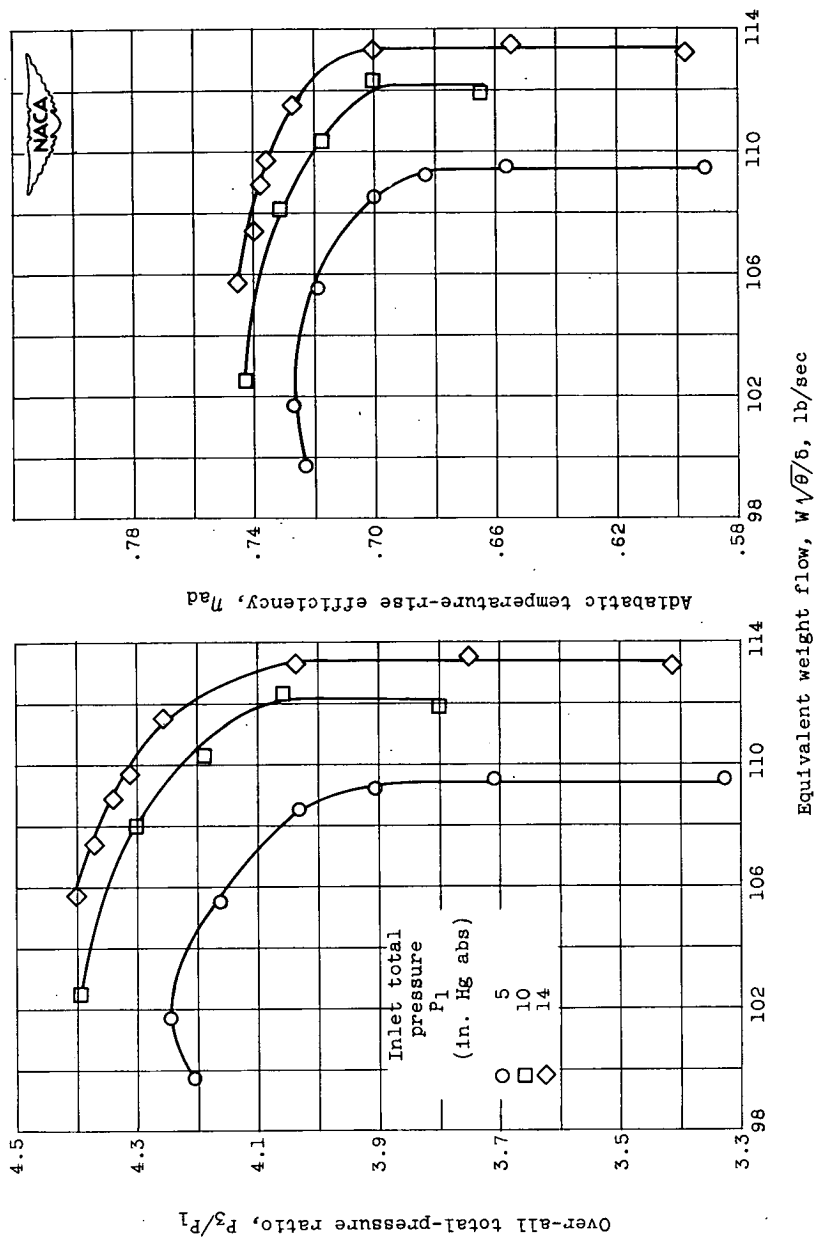


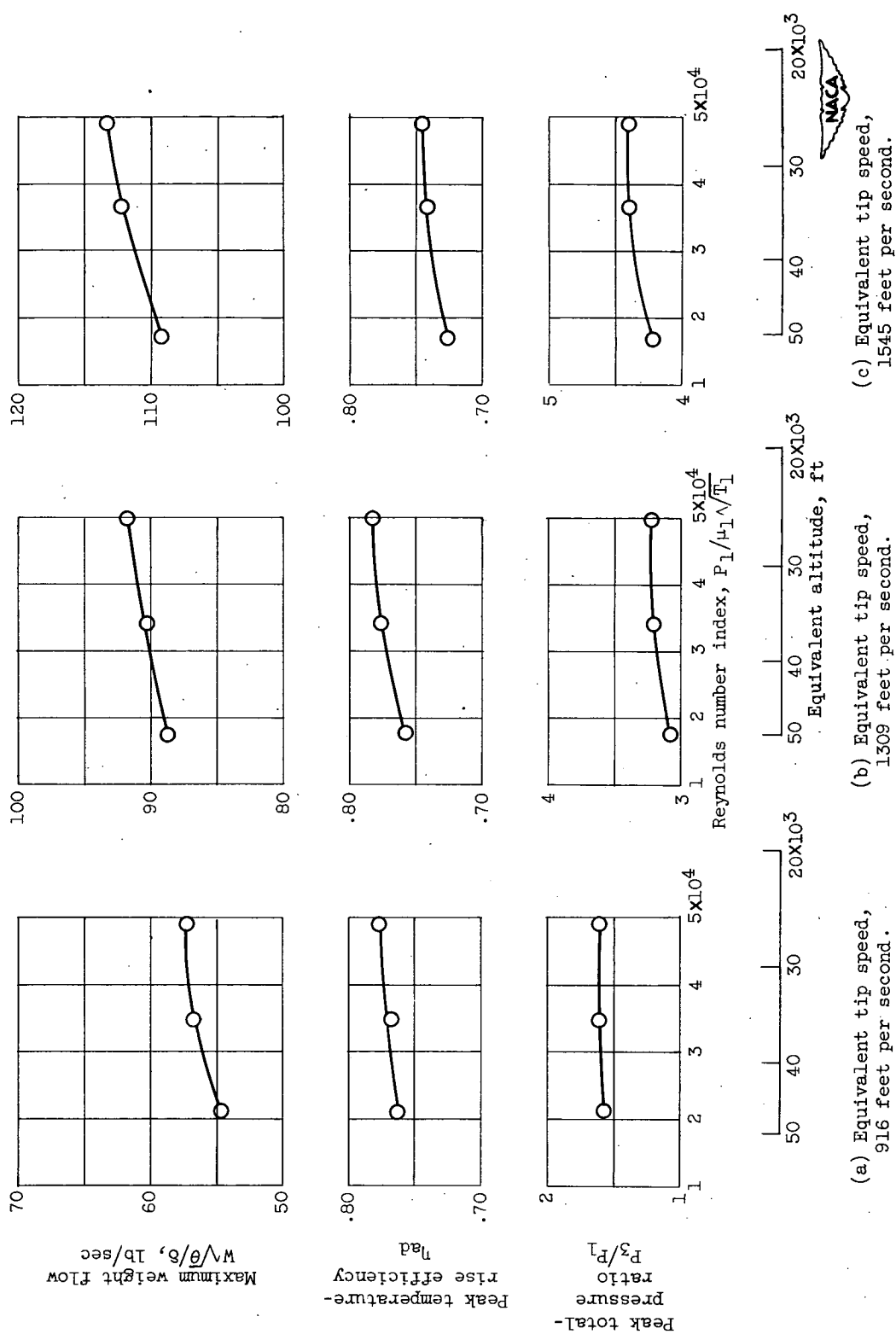
Figure 3. - Continued. Over-all performance of 23-blade compressor at four inlet pressures.



(e) Compressor pressure ratio at equivalent tip speed of 1545 feet per second.

(f) Compressor efficiency at equivalent tip speed of 1545 feet per second.

Figure 3. - Concluded. Over-all performance of 23-blade compressor at four inlet pressures.



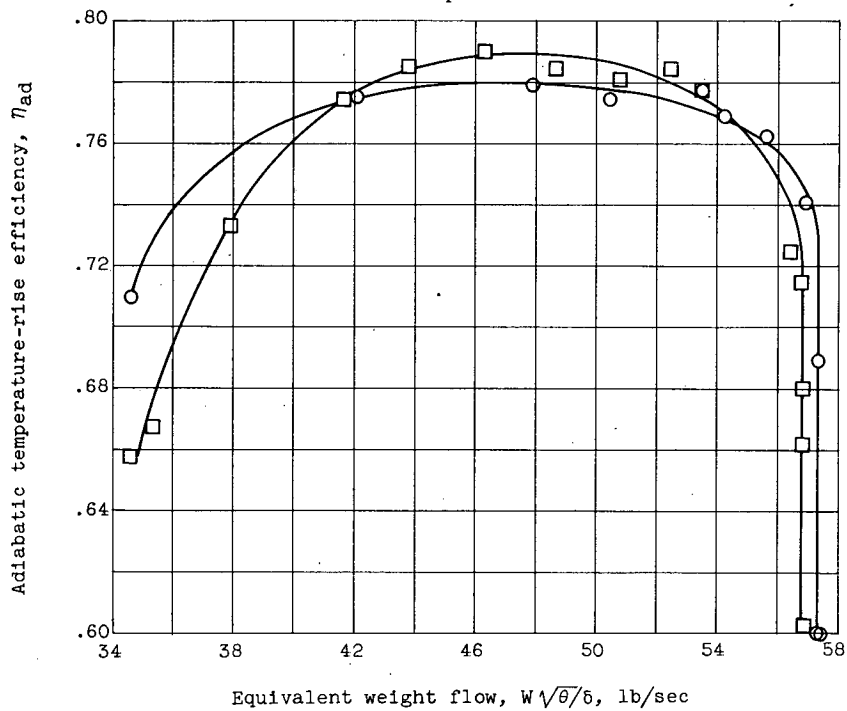
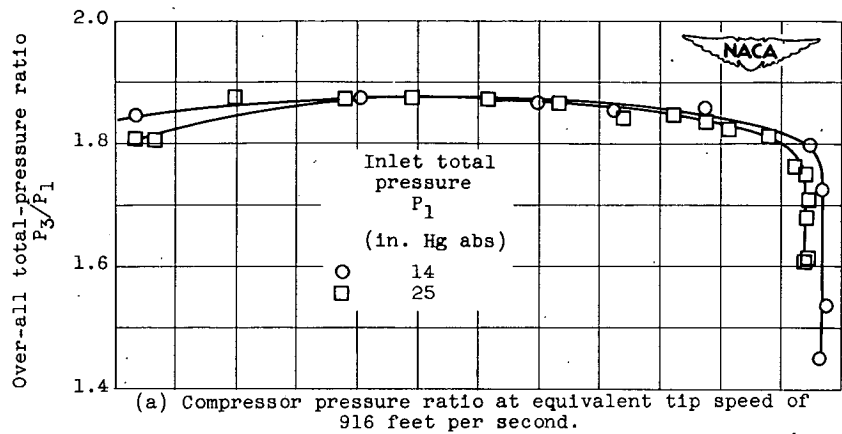
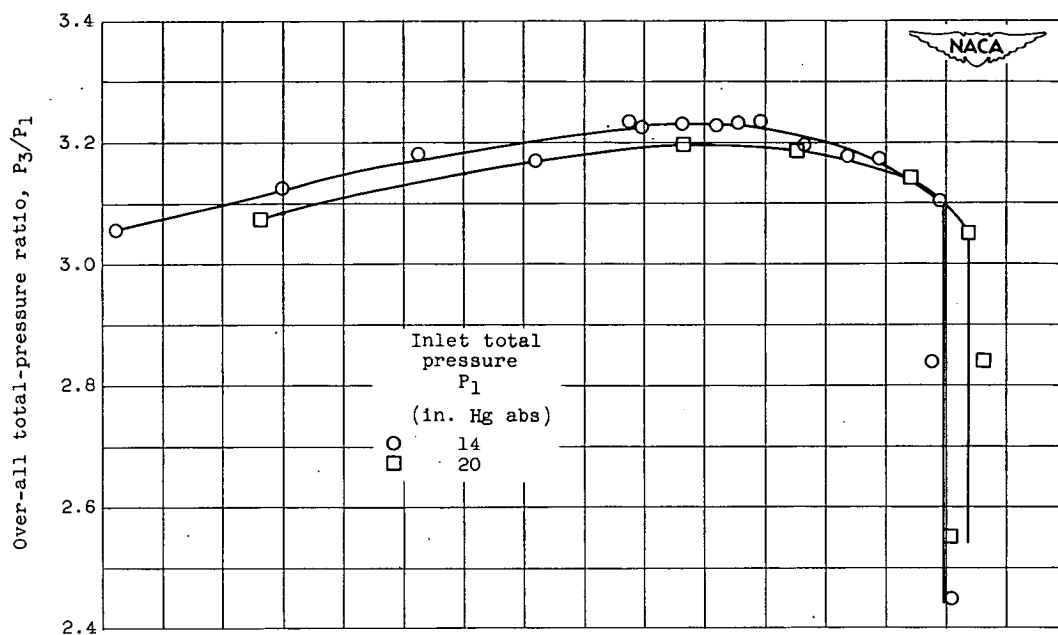
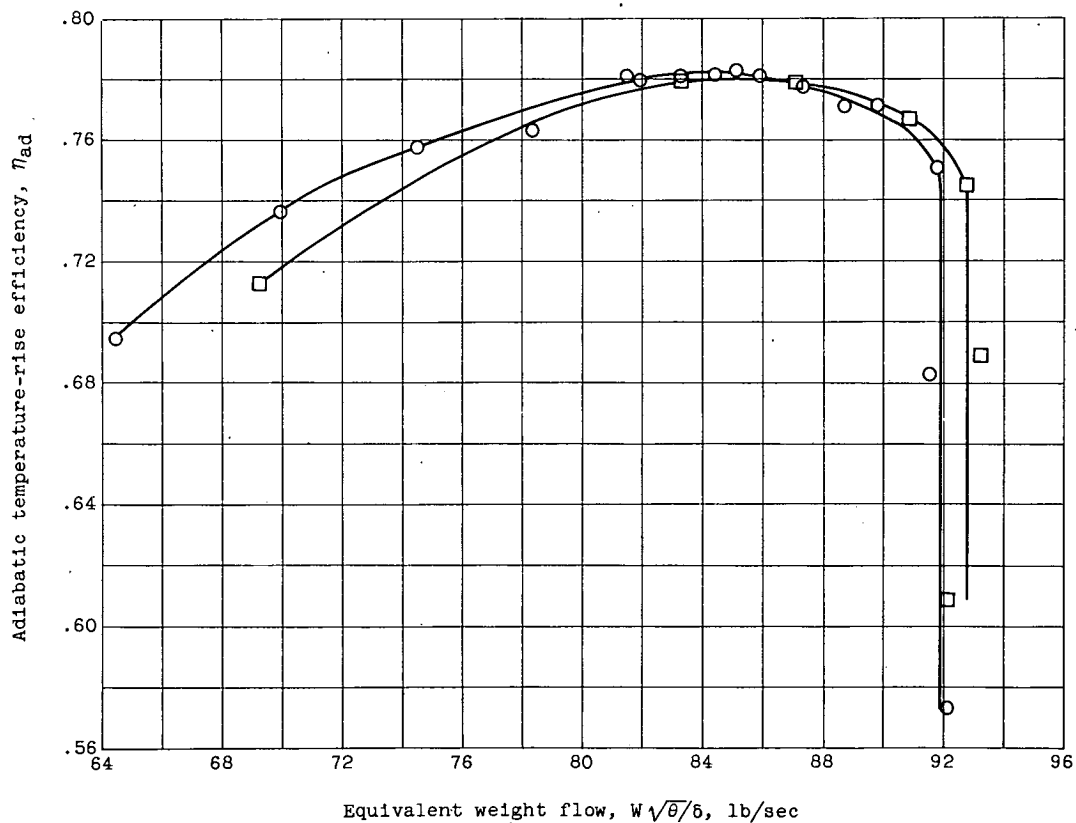


Figure 5. - Over-all performance of 23-blade compressor at inlet pressures of 14 inches of mercury and higher.

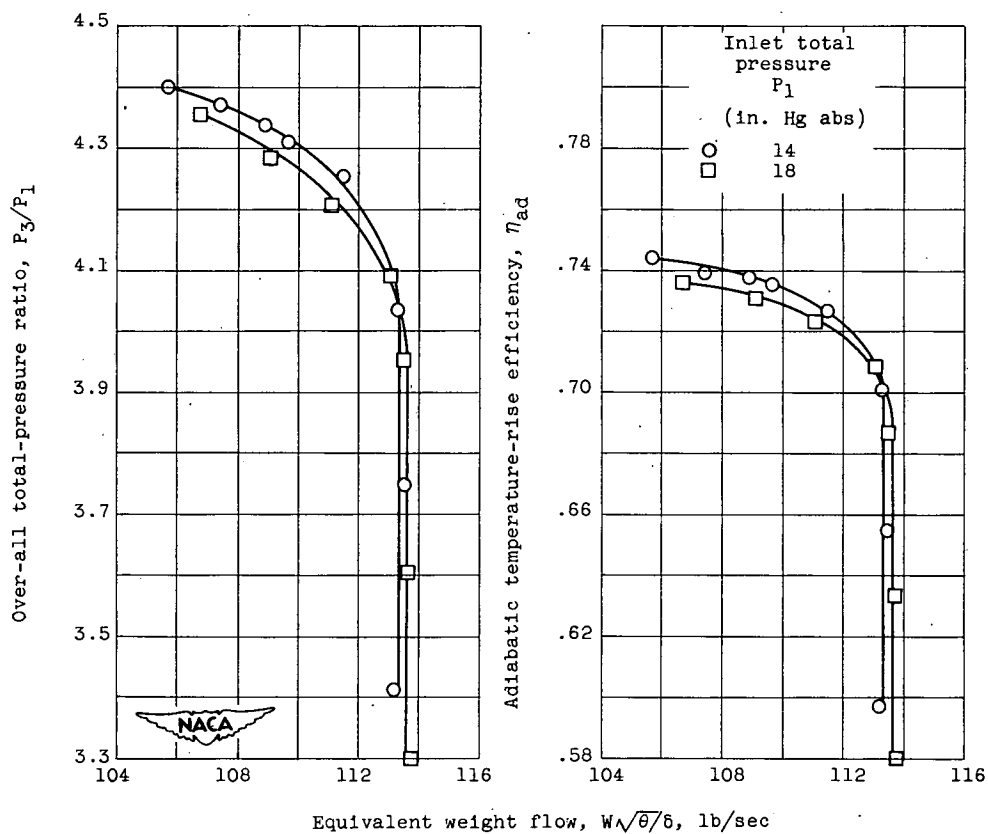


(c) Compressor pressure ratio at equivalent tip speed of 1309 feet per second.



(d) Compressor efficiency at equivalent tip speed of 1309 feet per second.

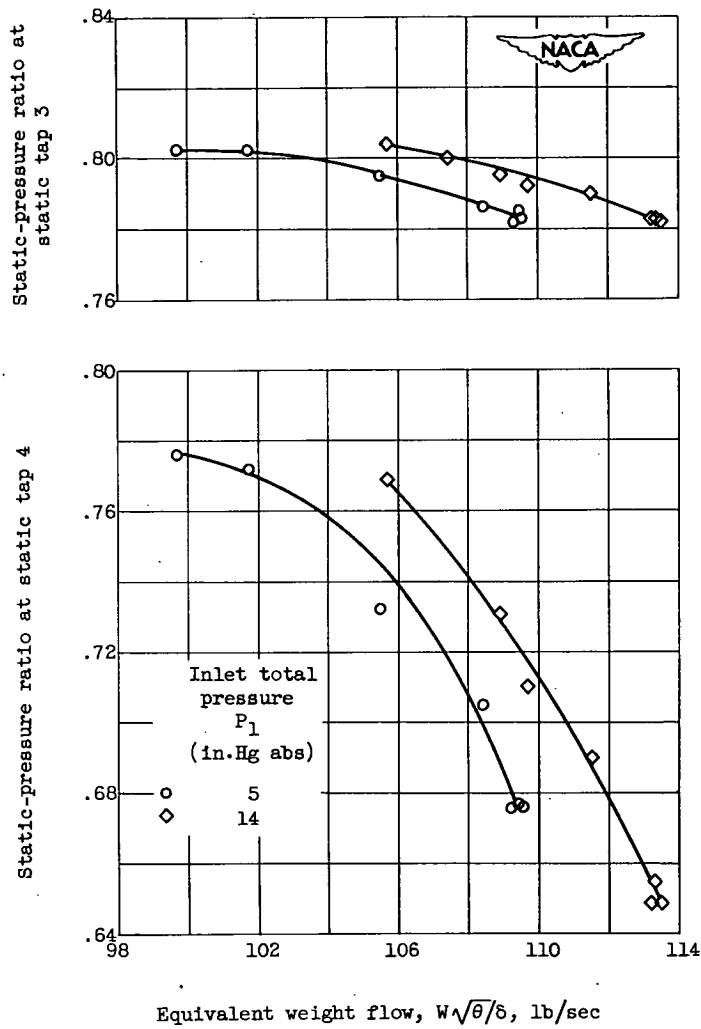
Figure 5. - Continued. Over-all performance of 23-blade compressor at inlet pressures of 14 inches of mercury and higher.



(e) Compressor pressure ratio at equivalent tip speed of 1545 feet per second.

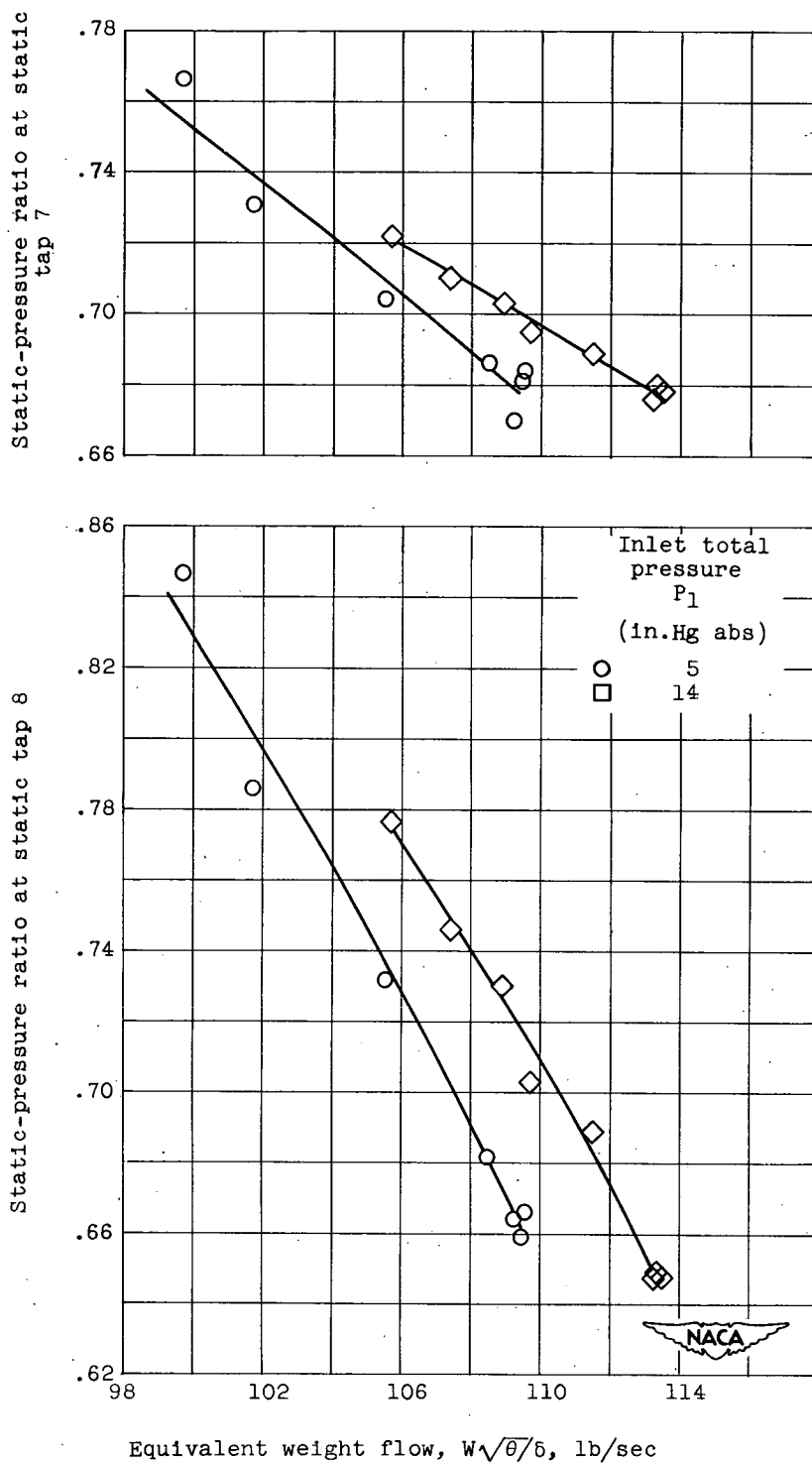
(f) Compressor efficiency at equivalent tip speed of 1545 feet per second.

Figure 5. - Concluded. Over-all performance of 23-blade compressor at inlet pressures of 14 inches of mercury and higher.



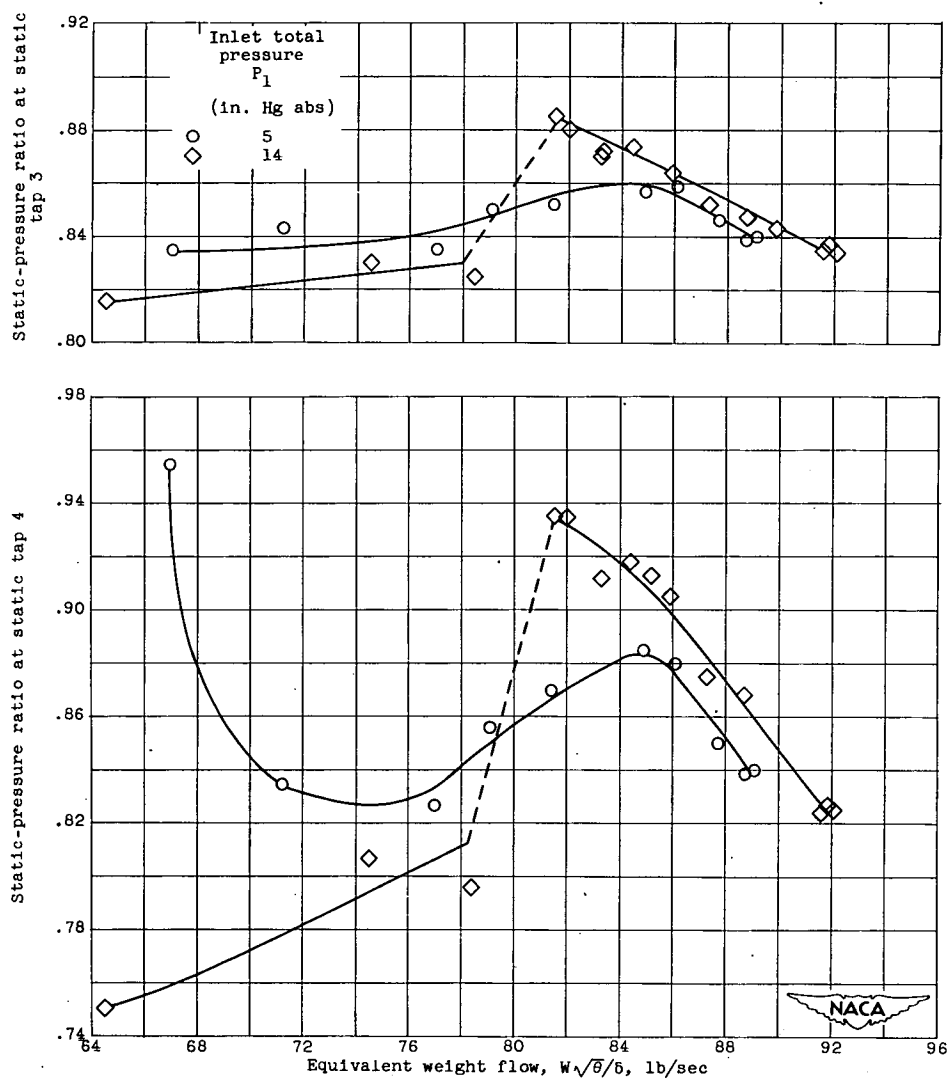
(a) Front inlet.

Figure 6. - Static-pressure ratios at inlets for equivalent tip speed of 1545 feet per second and two inlet pressures.



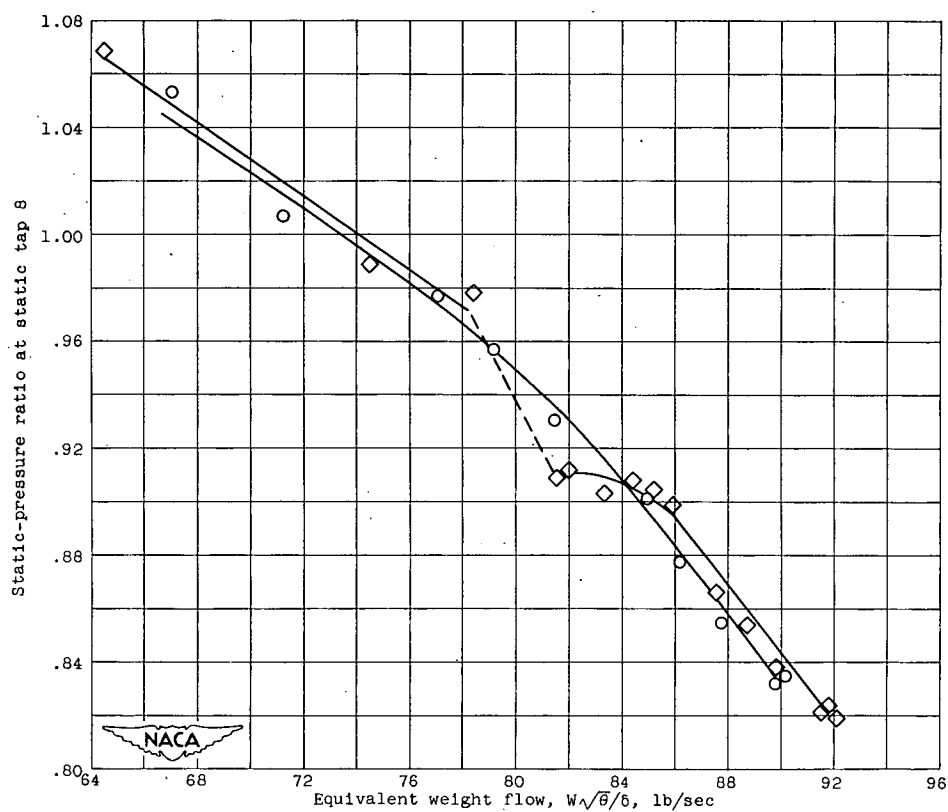
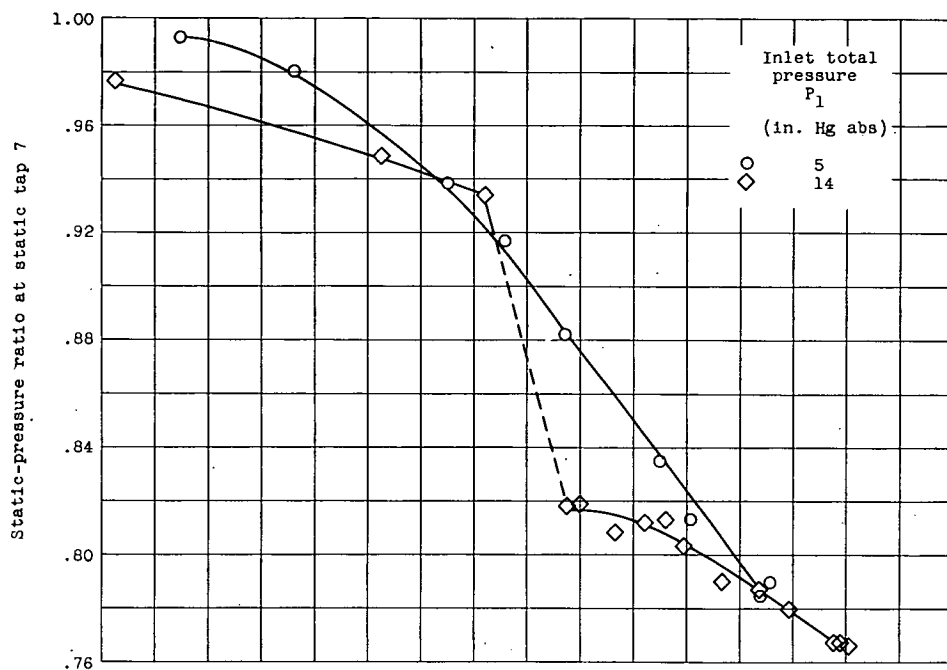
(b) Rear inlet.

Figure 6. - Concluded. Static-pressure ratios at inlets for equivalent tip speed of 1545 feet per second and two inlet pressures.



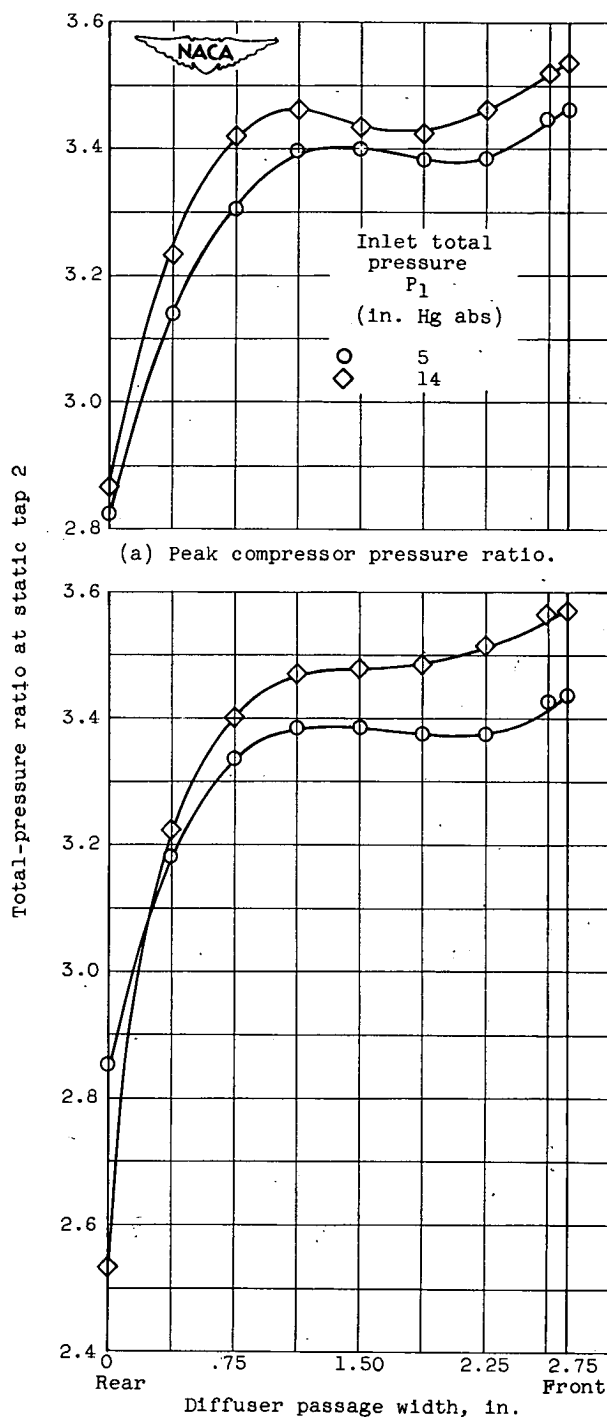
(a) Front inlet.

Figure 7. - Static pressure ratios at inlets for equivalent tip speed of 1309 feet per second and two inlet pressures.



(b) Rear inlet.

Figure 7. - Concluded. Static-pressure ratios at inlets for equivalent tip speed of 1309 feet per second and two inlet pressures.



(b) Constant equivalent weight flow corresponding to maximum at inlet pressure of 5 inches of mercury absolute.

Figure 8. - Total-pressure surveys at vaned-diffuser inlet for equivalent tip speed of 1309 feet per second.

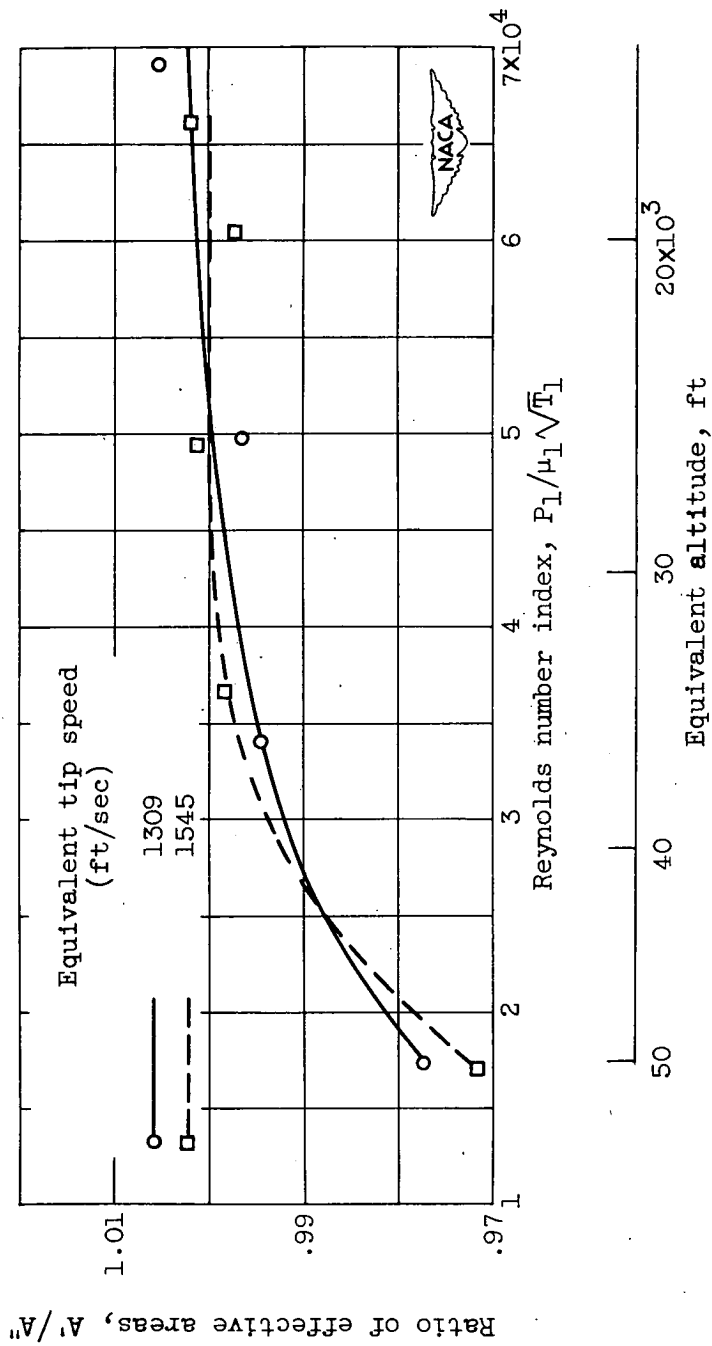


Figure 9. - Variation of effective area at choking point in vaned diffuser with Reynolds number index.